Defense Technical Information Center Compilation Part Notice

ADP013515

TITLE: Troubleshooting Vibration Problems. A Compilation of Case Histories

DISTRIBUTION: Approved for public release, distribution unlimited

This paper is part of the following report:

TITLE: New Frontiers in Integrated Diagnostics and Prognostics. Proceedings of the 55th Meeting of the Society for Machinery Failure Prevention Technology. Virginia Beach, Virginia, April 2 - 5, 2001

To order the complete compilation report, use: ADA412395

The following component part numbers comprise the compilation report:
ADP013477 thru ADP013516

TROUBLESHOOTING VIBRATION PROBLEMS
A COMPILATION OF CASE HISTORIES

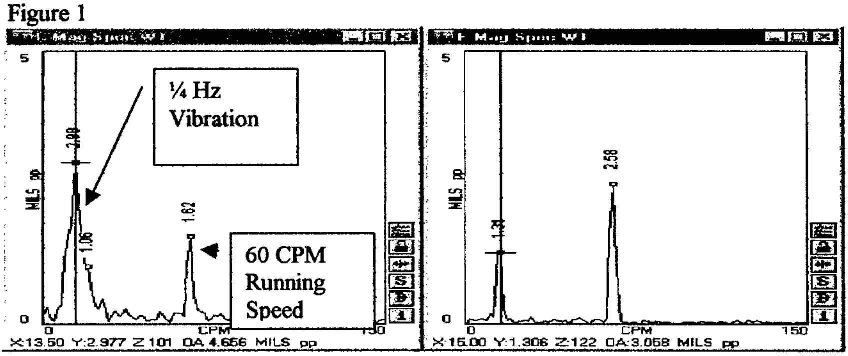
Nelson L. Baxter
ABM Technical Services, Inc.
8529 E. CR. 300 S
Plainfield, In. 46168
ABM@SURF-ICI.COM

ABSTRACT: Vibration analysis can be utilized in the solving both rotating and non-rotating equipment problems. This paper presents several case histories where vibration analysis was utilized to troubleshoot a wide range of problems.

<u>Key Words:</u> Analysis; fans; motor; pumps; turbines; vibration

CASE 1- ¼ Hz vibration in Hydroelectric Dam. During the operation of a hydro-electric turbine, the whole structure would begin to shake in a certain load range. The problem was identified as Rheingan's influence. Rheinghan's influence is caused by spiral vortex filaments that rotate at a speed lower than the turbine RPM. The load range at which the Rheingans influence occurred varied in this case, depending upon the up and down stream water level. The problem was that the vibration supervisory system did not register any excessive levels during the transient, so the operators who were located at a remote location, did not know what load ranges to avoid. The spectra shown in Figure 1 were taken from shaft proximity probes while the problem was present.

Figure 1



The above plots taken with a spectrum analyzer that was DC coupled showed that there was no difficulty in detecting the vibration. The reason that the supervisory did not detect the 15 cpm vibration caused by the Rheinghan effect was that its AC coupling capacitor filtered out the vibration. Since nothing can be done to prevent the Rheinghan effect, modifications to the supervisory system are being considered to allow the operators to at least avoid the unstable load ranges.

CASE NO. 2- VIBRATION OF STEEL STRIP IN STEEL MILL CAUSED BY INDUCTION FURNACE.

EQUIPMENT: An induction furnace in a steel mill was used to heat and diffuse galvanize into the steel strips.

SYMPTOMS: During the induction process, a loud high pitch frequency would radiate from the steel plate. When the sound would begin, vertical stripes would also appear on the plating. The stripes were causing the steel to be rejected by the customer of the steel mill.

TEST DATA AND OBSERVATIONS: An FFT analyzer was set up to determine the frequency of the sound, along with the vibration on the induction furnace and the current being supplied to the induction coils. The frequency being detected in all three cases was at 7250 cycles/second. This frequency corresponded to the operating frequency of the induction furnace. To determine if a change in frequency would have an effect on the problem, the furnace frequency was increased to 9000 Hz. The stripes did not disappear at the higher frequency, but merely moved closer together

CONCLUSIONS AND RECOMMENDATIONS: It was determined that the induction furnace was exciting the natural frequencies of the plate, creating standing waves, which resulted in the stripes being formed. Since the thin plate had several natural frequencies within the normal operating range of the furnace, changing from one frequency to another did not help. Increasing the frequency made the stripes closer together, decreasing the frequency resulted in the stripes being further apart. When turbine generators are brought to operating speed, there are several speeds which match the natural frequencies of the various blade lengths. Due to thermal stress considerations, it is necessary to operate in the range of speeds where problems can occur, while the rotors get thermally stabilized. To prevent damage from occurring to the blades, the speed on the turbine is continually varied, rather than staying at one speed, during the thermal soak periods.The above analogy was used to approach the striping problem. Rather than operating at one frequency, as it had in its past, the induction furnace's control circuit was designed to continuously vary the frequency . When this modification was made, the striping problem was eliminated.

CASE NO. 3 VIBRATION OF MICROSCOPE IN MICROSURGURY ROOM OF A HOSPITAL

Equipment: A microscope mounted from the ceiling of a surgery room. The special microscope was used by the surgeon during operations which involved replanting limbs.

Symptoms: The chief surgeon complained that the image was jittery and that it was very tiring to operate under those conditions, particularly when the scope was set for its maximum magnification.

Test Data and Observations: The scope was set to its greatest magnification and printed material was placed on the operating table. Vibration was clearly noticeable, just as the surgeon had indicated. Vibration spectra were taken on both the table and the microscope. The levels on the table were very low across the spectrum. However, the levels on the microscope were significant. A view of the vibration spectra revealed that peaks were present at 225 CPM and 435 CPM. To trace the source of the vibration,

levels were measured on the top of the microscope's isolator and on the structural steel supporting the isolator . It was discovered that the levels on the isolator were seven times higher than on the steel support. This meant that instead of isolating the microscope from the structural vibration, the isolators were actually amplifying the vibration which was present on the I-beam. To determine the cause of the amplification, an impact test was performed on the microscope to determine its natural frequencies. It was found that the natural frequency of the scope on its isolation system matched the vibration which was present on the I-beam.

Conclusion and Recommendations: Isolators perform their isolation function by creating a system with a natural frequency tuned much lower than the expected disturbing frequency. This in turn creates a mechanical low pass filter, which will not pass the higher frequencies. A problem can, however, occur if a low frequency is present near the low tuned natural frequency of the isolated system. Instead of isolating the frequency, the isolators will then actually amplify the levels. The solution in this case was to ground out the isolators. When the isolators were grounded out, the levels on the scope dropped to acceptable levels. The frequencies which had been present were due to isolators on the roof fans being tuned to the same frequencies as the microscope. Flow excitation in the fans excited the fans' isolated natural frequencies, which were transmitted through the structural steel. Grounding of isolators should only be tried if nothing else works. When the isolators are grounded, higher frequencies, if any are present, will obviously pass through. In this particular case, grounding out the isolators didn't introduce any significant higher frequency vibrations.

CASE NO. 4- TORSIONAL VIBRATION ON A RECIPROCATING PUMP

Equipment: 66 RPM Reciprocating Water Pump driven by a gear case and belt reduction.

Problem: Excessive torsional vibration at the pump speed was being picked up at the gearcase.

Background on Torsional Testing:

Torsional testing generally is performed using either one of two methods. The first is the use of a strain gauge to measure the alternating torsional strain. The second method involves measuring the change in the passing frequency of equally spaced gear teeth or equally spaced reference marks. The change in passing frequency of equally spaced marks on a shaft is an indication of corresponding changes of angular velocity. This data can therefore be integrated to produce angular displacement.

Test Data and Observations:

For this test, both the strain gauge and equally spaced reference mark techniques were used so that a comparison between the two methods could be made. A strain gauge was mounted at a 45 degree angle on the drive shaft between the gearcase and the belt driving the reciprocating pump. An FM transmitter and a battery were also mounted on the drive shaft to transmit the strain information to an antenna and FM receiver-demodulator.

The above set up was calibrated by putting one end of the drive shaft in a vice and applying 100 ft-lb of torque to the other end. While the 100 ft-lb of torque was applied,

the output of the demodulator was measured with a volt meter. The calibration constant from this test was then input into an FFT analyzer, resulting in readings directly in ft-lbs at each frequency present. The second test involved putting equally spaced photo-reflective tapes at twenty locations around the output hub of the gearcase. A photocell device was then mounted to pick up the pulse train of the reflective tapes. The output from the photocell was then input to a torsional-demodulator-integrator which produced an output of 200 mv/degree peak to peak. With the above combination, it was therefore possible to measure the torque being fed back to the gearcase from the pump and the amount of angular displacement it produced. When the pump was operating against no appreciable back pressure, there was 1.5 degree of angular displacement present at the gearcase hub at 66 cycles/minute. The torque from the strain gauge at this condition was 27 Ft-lb. As the back pressure on the pump was increased, the above values also increased. When the output pressure from the pump was 180 PSI, the torsional vibration had increased to 8.79 degrees and the alternating torque to 123 ft-lb, both at the pump speed of 66 cycles/minute.

Corrective Action:
The above data showed that the alternating torque peaks from the pump were too high. To correct this problem, a flywheel was added to level out the torque peaks by absorbing energy during one half of the cycle and return it to the system on the other half cycle.

Final Results:
The alternating torque values were reduced by a factor of three by the addition of the flywheel.

Conclusions and Recommendations:
The above case history shows how two different techniques led to the same conclusion. The test method used will depend on what the investigator needs to know, the test equipment available and the accessibility of the machinery to be tested. Another interesting note concerning the above is that when the coherence was measured between the two signals with a dual channel analyzer the level was .98. This indicates a direct correlation between the alternating torque on the drive shaft and the displacement on the gearcase hub, which is as would be expected.

CASE No. 5- GHOSTS
Symptoms of Problem: In a small Midwestern town, the residents complained that they felt movement of their houses, particularly at night. One of the residents stated that her sister would no longer come and visit her because she thought the house was haunted with ghosts. She felt this way because lamp shades would move, pictures would rattle and rocking chairs would rock without anyone being in them.

Test Procedure: In an attempt to identify the cause of the above problem, vibration measurements were made at several of the houses in the community, along the sidewalks and at a factory located near the houses. The testing was performed with an SD-380 analyzer utilizing a 1000 mv/g low frequency seismic accelerometer to convert the mechanical motion into an electronic signal.

Test Results:
The vibration signature taken at one of the houses where complaints had been registered showed a level of 1.22 mils at 300 cpm. It was observed that the amplitude of the vibration would oscillate, indicating that more than one frequency might be present. (Beat frequencies are generally produced by two or more closely spaced frequencies adding and subtracting as they go in and out of phase.) To determine if there was more than one frequency present, the zoom feature of the SD-380 analyzer was utilized. A 16:1 zoom plot of the vibration at one of the residences clearly showed the presence of several frequencies, all close to 300 CPM. The next step in the investigation was to make a survey of the vibration present at the nearby factory. The factory was a foundry, which contained several vibratory conveyors that moved parts from one area to another. Zoom plots were taken at each conveyor to determine their frequencies as close as possible. Exact matches were found between the frequencies present at the houses and several of the vibratory conveyors in the factory. Additional testing confirmed the correlation between the operation of the conveyors and the vibration at the houses.

Conclusions: Rather than having ghosts, the residents of the small town were experiencing low frequency vibrations from the vibratory conveyors in a nearby factory. The 5 hz (300 cpm) frequency is easily perceived by the human body, particularly at night when other motion and noise is at a minimum. It also can excite low stiffness structures (i.e. lampshades and rocking chairs).

CASE NO. 6

VIBRATION OF NUCLEAR MAGNETIC RESONANCE MACHINE

Equipment: Nuclear Magnetic Resonance instrumeny used to test chemical samples.

Symptoms: Following the movement of the unit from a second floor location to a third floor room, the NMR instrument performed poorly. A test technician determined that everything was operating properly within the unit. The technician thought that vibration might be the cause of the problem, so tests were performed.

Test data and Observations: A signature taken at the probe of the NMR unit in the new location, showed a level of 8,190 micro g's. The floor beside the NMR unit had a level at the same frequency of 1,550 micro g's. The readings meant that the vibration on the NMR unit was 5.2 times higher than the level measured on the floor at the 26 Hz frequency of the vibration. To determine the cause of the amplification, a resonance check was made. The floor was impacted and the response was measured on the detector of the NMR unit. The transfer function clearly showed a peak at 26 Hz, indicating that the Unit was resonant at the frequency which was present on the floor.

Conclusions and Recommendations: It was concluded that fans in an HVAC room near the third floor location were providing the 26 Hz forcing function. The resonant condition was significantly amplifying the vibration that was present. It was recommended that the NMR unit be installed on isolators with a 95% efficiency in attenuating the 26

Hz vibration. The 95% reduction resulted in levels which were lower than those the unit had been exposed to in its original location, where it had operated satisfactorily. Following the installation of the isolators, the NMR unit performed well.

CASE NO. 7-VIBRATION INDUCED BY SOUND

Equipment: Rotary casting conveyor in a foundry.

Symptoms: After the installation of a rotary casting conveyor, vibration of the walls and particularly on the windows in the control room of the foundry experienced high levels of vibration.

Test Data and Observations: The locations with the highest levels of vibration were the windows in the control room. A plot of the vibration measured on the foundry windows showed a level of 39.2 mils near the center of one of the windows. A frequency of 885 cycles per minute was predominant in the spectrum. The 885 CPM vibration on the windows was also found to be present on the walls of all the offices in the foundry. This frequency matched the frequency of a large rotary casting conveyor. Vibration measurements next to the conveyor were, however, low. The conveyor was mounted on springs and also was fitted with dynamic absorbers, which were apparently working as designed, considering the low levels observed on the floor next to the conveyor. The next test involved taking measurements with a microphone. The output from the microphone was analyzed on an FFT analyzer and it was found that the sound level at 885 CPM (14.75 Hz) was over 100 dB. Since this was below the normal hearing range for humans, the sound level did not seem bad, however, it could be felt and a sheet of paper held in front of the conveyor would move noticeably. The final test involved performing a resonance check on the window. A plot of the response of one of the control room windows showed that the natural frequency of the window was very close to the frequency of the pressure waves being emitted by the rotary casting conveyor.

Conclusions and Recommendations: The vibration problem undoubtedly was caused by the rotary conveyor. The transmission path was through the air rather than through the structure. The windows being resonant near the operating frequency of the conveyor were further amplifying the problem. It was recommended that the windows be fitted with cross braces to move their natural frequencies away from the operating frequency of the conveyor and that a sound absorbing enclosure also be built around the conveyor.

CASE 8 - HIGH ALMOST 2X VIBRATION ON A HIGH PRESSURE CORE INJECTION PUMP AT A NUCLEAR PLANT

Equipment: Steam Turbine Driven High Pressure Core Injection Pump on a Boiling Water Nuclear Reactor.

Symptoms: Operators of Nuclear Power Plants are required to comply with section XI of the ASME code regarding In service Inspection. The code states that base line readings are to be taken in Mils Displacement and that if the base line readings double, then action

must be taken. On this particular pump, the base line level had nearly doubled, so an investigation was initiated

Test Data and Observations:

The unit consisted of a turbine driver, a high pressure pump, a gear case and a low pressure booster pump. Vibration spectra were obtained for all the bearing locations on the pump train. The readings were at normal levels everywhere except on the High Pressure pump in the horizontal direction. A level of 1.4 inches per second was present at what appeared to be twice the running speed of the unit at that location. A cascade plot of vibration on the HP pump in the horizontal direction from the inboard end to the outboard end showed that the vibration was high at each end of the pump, but was nearly zero in the center. Due to the large difference between the vertical and horizontal readings and the presence of an apparent rigid body pivoting mode, a horizontal resonance was suspected. An impact test was performed on the pump. A natural frequency at near twice running speed was found. This mode matched the response found while the pump was running (i.e. high on the ends and low in the middle). Since the vibration was predominately at what appeared to be twice running speed, it was suspected that misalignment might be the source that was exciting the horizontal structural resonance. The alignment was checked and found to be out of specifications. The alignment was corrected and a test was run on the unit. There was no improvement in the level of vibration. In fact it was slightly higher on the test run than it had been previously. This change in the level of vibration followed a pattern that showed that the vibration in the 2X cell would vary from .9 to 1.6 in/second from one test run to another. In an attempt to determine the phasing of the vibration, a once per revolution pulse was used as a reference trigger for the FFT analyzer and a signature was taken in the synchronous time average mode. As the number of averages increased, the vibration at what appeared to be twice running speed disappeared. This was one of the breakthroughs in the analysis of the problem. It meant that the vibration was not phase locked to the high pressure pump shaft.

The drawings were again examined on the pump and it was found that the gearcase had a 1.987:1 reduction. In addition, it was discovered that the low pressure pump had a four vane impeller. The pieces were beginning to fall together. The pump manufacturer was contacted with the above information. The pump man recalled that there had once been a case of an acoustical resonance with a similar pump. To determine if this could be contributing to the problem, the piping between the low pressure and high pressure pump was measured. It was found that the length of pipe connecting the pumps was equal to one half wave length of the low pressure pump blade pass frequency. Since the low pressure pump had 4 vanes and the gear reducer had a 1.987:1 reduction, the vane pass frequency looked exactly like twice the running speed of the High Pressure Pump. As a final test to confirm the above theory, a tach pulse was put on the low pressure shaft. The vibration pickup was placed on the high pressure pump. The vibration on the HP pump was found to indeed be phase locked to the low pressure pump.

Conclusions and Recommendations:

The problem on the High Pressure pump was not at twice the HP pump running speed, as it had appeared. It was actually at the vane pass frequency of the low pressure pump. It appeared to be at twice the HP pump running speed because of the 4 vane impeller in the LP pump and the almost exact 2:1 speed reduction between the two pumps. The problem was amplified by the acoustical resonance of the pipe connecting the discharge of the LP pump to the suction of the HP pump. The vibration was further amplified by the horizontal structural resonance of the HP pump casing. The major clue to the solution of the problem was that the vibration on the HP pump was not phase locked to the shaft on that pump and was therefore coming from another source. The final clue was the past acoustical resonance problem encountered by the pump manufacturer. The recommenced solution was to change the 4 vane impeller out to a 5 vane design. This change entirely solved the problem.

CASE NO. 9-MACHINE TOOL VIBRATION

Equipment: Machine that cut chamfer on wrist pin hole of rods for automotive engines.

Symptoms: A loud high pitch sound was produced during the backside chamfer operation. Examination showed that the surface that was machined was very rough and completely unacceptable to the customer.

Test Data and Observations: The manufacturer indicated that the problem might be due to bearings or gearing within the chamfer head. The test data proved that this was not the case. During the cutting operation, a spectrum was taken on the machine being used to cut the chamfer, while it was operating at 900 RPM. The data was taken on the chamfer head in the vertical direction. The frequency of the vibration was at approximately 115,000 cycles per minute (1916 Hz). Data was then captured in the digital buffer of a spectrum analyzer during the transient. It could be determined from the captured data that the vibration would build up, then abruptly quit when the cut was completed. The peak level of vibration measured on the part being machined, reached a level of .82 inches/second. Another test was performed with the machine operating at 1200 RPM, which was a 33% increase in speed. The vibration frequency which was produced on the machine and the part was the same frequency which had been present when the machine operated at 1200 RPM. Based upon the fact that the frequency did not change when the tool RPM was varied, it was suspected that a natural frequency was being excited. To test this theory, an impact test was performed on the cutting tool, which was the most likely source of the problem. A resonance peak clearly showed up at 114,750 CPM. A mode shape of the first natural frequency of the tool was produced by profiling the imaginary components of the transfer functions taken along the tool's surface . The 114,750 CPM mode was found to be the first cantilever mode of the cutting tool. To complicate matters, it was discovered that the part in its holder also had a natural frequency near that of the tool.

Conclusions and Recommendations: It was recommended that the tool be modified to separate its natural frequency away from that of the part. It was also recommended that

the basic process be reviewed. Since the problem only occurred when the back side cut was made, the problem may have originated from trying to make the cut when the tooling bar was under tension rather than in compression.

CASE 10- EXTREMELY HIGH LEVELS OF VIBRATION ON THE END CAP OF A LARGE PIPE.

SYMPTOMS: A large pipe (37" DIAMETER)at a refinery had very high levels of vibration on an end cap after an expansion joint. The levels were over 4.0 inches/second. There had been failure of several of the retaining rods which spanned an expansion joint.

TEST DATA AND OBSERVATIONS: A vibration spectrum was taken on the end cap. The vibration was found to occur at 4500 cycles/minute with a level of 4.47 inches/second. The area surrounding the pipe was checked for rotating equipment operating at that frequency. No machinery was found which operated anywhere close to that speed. A visual exam of the pipe showed that there was a large butterfly valve up stream of the end cap. The end cap was on a dead end section of a tee. Conversations were held with plant personnel to determine when the vibration had started and what, if anything, had been done to the piping prior to the high vibration and retaining rod failures. The first response was that work had been performed on the expansion joint, however, "nothing had been changed". Further discussions and examination of the piping drawings did show that one thing had indeed been changed. A baffle had been removed just upstream of the expansion joint. The baffle was a thick flat plate with two small holes in it. The plant personnel didn't think that it served any purpose. An overall review of the system showed that it was very important. What was occurring was that the butterfly valve was causing flow disturbances. The valve was found to be operating only 30% open. This resulted in pressure pulsations in the pipe. The baffle was acting like a low pass filter which allowed the static pressure to equalize, but would not let the dynamic pressure pulsations pass. The pressure pulsations were probably small, however, there were two design features that caused the vibration levels to be high. The first was the amount of area on the end cap. A 37" diameter end cap has 1075 square inches of surface area. Therefore even a small pressure pulsation can generate a large force, when acting on such a large area. The second reason was the length of the retaining rods. The retaining rods were 20 feet in length, so the stiffness value was low.

CONCLUSIONS AND RECOMMENDATIONS: The vibration problem was the result of pressure pulsations within the pipe acting on a large area with low stiffness. The removal of the baffle had been a key element in the problem. The plant was advised to install short bolts across the expansion joint until the unit was brought down for its next outage. The purpose of this action was to stiffen up the system and provide a backup to the long bolts that had been failing. This action was possible because the piping was already at its maximum temperature. During the following outage, the baffle was reinstalled in the pipe and the short bolts across the expansion joint were removed. The vibration problem was eliminated.

CASE 11-LARGE ALIGNMENT CHANGE ON BOILER FEED PUMP CAUSING FAILURE OF PUMP AND CRACKING OF TURBINE PEDESTAL.

EQUIPMENT: Boiler Feed Pump on a 500 Megawatt turbine generator.

SYMPTOMS: After being in operation only a few months, the boiler feed pump on at a new generating station failed. The inboard seals were wiped out and the stage next to the coupling destroyed. Due to the type of failure, the plant initiated an alignment study.

OBSERVATIONS AND TEST RESULTS: In order to determine the amount of movement from the hot to cold condition, Dynalign bars were mounted between the pump and the turbine driver, while the unit was in operation, following repairs. When the unit was brought off line, there were no significant changes recorded. However, a few minutes later, the Dynalign system was found to be entirely out of range. When the operators were questioned as to what had happened during that period, they replied that the only thing they had done was to break vacuum on the main turbine. To determine if the vacuum had anything to do with the apparent alignment change, the bars were reset and long range probes were installed. The vacuum was then reapplied to the system. Everything appeared normal until the vacuum reached 11" of Hg. At that point, the changes in the gap voltages started to be recorded. By the time full vacuum was achieved, the relative motion between the turbine and the pump was over .100". The vacuum was then released and the readings moved .100" in the opposite direction. The test was repeated with identical results. An examination of the system was then performed. The boiler feed pump turbine was found to be connected to the main turbine condenser by a large pipe. On the end of the pipe was an end cap. Three expansion joints were installed between the main condenser and the feed pump turbine. The purpose of the expansion joints was to isolate the boiler feed pump turbine from stresses induced by thermal growth of the condenser pipe. Thrust canceling rods were installed between the end cap and the main condenser. The thrust canceling struts transmitted the atmospheric load on the end cap to the condenser. The source of the problem was that threaded studs in the thrust canceling struts were sliding into the struts. This was the result of the welds on the large nuts on the back side of the plates the ends of the struts failing. The net result of the failure was that atmospheric pressure which was being applied to the 6' diameter end cap was pushing on a 20' vertical run of pipe attached to the bottom of the feed pump turbine. The large force applied to the 20' lever had the capability of generating nearly 1 million ft. lbs of torque to the turbine. Examination of the concrete turbine base showed that it had several cracks from the bending torque. The thrust canceling struts were repaired and the alignment changes were rechecked as vacuum was reapplied. Following the repair, there were no significant changes in alignment as a result in variations in vacuum.

CASE NO. 12- Large fan that could not be balanced.

Equipment- A 5000 HP 720 RPM fan at a power plant.

Symptoms: Above normal levels of unbalance. Several attempts were made to balance the unit, all of which were unsuccessful.

Test Results: The casing vibration levels were around 2-3 mils. Since there was difficulty in balancing the fan, shaft stick measurements were taken to determine the absolute motion of the shaft. The shaft movement was discovered to be over 17 mils. Since the bearing clearance was only 8 mils, there was a strong indication that the bearing was moving in the housing. A large plunger bolt at the top of the bearing was tightened. After tightening, the casing vibration increased to 21.5 mils. The fan was then easily balanced to below 1 mil. What had been occurring was that with the bearing moving in the housing a non-linear system was present. This made balancing all but impossible.

CASE NO. 13- Incorrect selection of isolators.

Equipment: Several air handler units

Symptoms and test results: A number of air handler units in a large city all had the same symptoms. The fans operated with acceptable levels of vibration, however, the motor drivers all had high levels of vibration in the vertical direction. When the spectra were examined, it was discovered that the primary component of the motor vibration was at the fan's operating speed. An investigation resulted in finding that the isolators under the motor were sized improperly. The isolators were too stiff. This resulted in the natural frequency of the isolated system matching the operating speed of the fans. The isolators were therefore acting as amplifiers of the fan vibration rather than isolators of the motor vibration. What appeared to have occurred is that the same size isolators were used for the motor as were used for the heavier fans.

Case No. 14- High axial vibration on a fan due to a disk wobble natural frequency.

Equipment: A belt driven exhaust fan operating 1200 RPM

Symptoms and test results: The fan's axial vibration was always high. The fan would be balanced and then a couple of weeks later, the axial vibration would go back up. Due to the sensitivity of the fan to unbalance, a resonance was suspected, so natural frequency tests were performed. There was no natural frequency match when the fan was struck in a lateral direction. However, when the fan was impacted axially, there was a match with running speed. To gather more information, the mode shape was measured. It was found that the shaft was the node point and that the opposite sides of the fan were out of phase. This is commonly called a disk wobble natural frequency. Fans that have this problem commonly exhibit sensitivity to unbalance, particularly in the axial direction.

The solution in this case was to simply change the fan's speed. If this is not an option, then stiffening of the back plate of the fan wheel may be necessary.

CASE NO. 15- Cavitation destroying impellers on large circulating water pumps.

Equipment: Large low RPM 156,000 gpm circulating water pumps.

Symptoms: The impellers on the circulating water pumps on a cooling lake at a large power plant were failing. The failure mode was cavitation. The impellers looked like they had been attacked by metal eating termites. When a vibration spectrum was taken, the spectrum contained a large amount of broad band energy with no distinct peaks.
The key to the analysis, as is the case with a good percentage of pump problems was to look at the flow head curve. The flow head curve indicated that at the design flow of 156,000 gallons per minute that the back pressure would be 30 ft. When the back pressure was measured, it was only 10 ft. What had happened was that during cold weather when the lake was cold, operations was operating with only one pump to reduce power consumption. The system was designed to operate against the back pressure produced by two pumps in parallel. When only one pump was on, the system back pressure dropped and the one pump that was on line went into cavitation.

CASE NO. 16- Low pump flow destroying antifriction bearings in a pump.

Equipment: Double suction single stage pump

Symptoms and test results: Three identical pumps sat in a row at a power plant. The bearings were failing on one of the pumps every few months. The other two pumps had no failures. Alignment was checked and different bearings were tried. Nothing helped.
While the pump was in operation, it was noticed that the shaft vibrated in the axial direction. This is called axial shuttling. It can occur when a pump is operating against too much back pressure. The suction and discharge pressure were therefore measured and compared to the flow head curve. It was discovered that the pump was operating at its shutoff head. Based upon this observation, the system was examined. The pump in question, pumped water from a tank in the basement up seven stories to another tank. The tank on the upper floor had a level switch that shut a control valve when it was full. When this occurred, water from the pump flowed through a bypass line back to the tank in the basement. The flow through this recirculation line was controlled with an orifice plate. The orifice plate said that it had a 2" hole, however, specifications said that the hole should have been 3". The surprising thing that was discovered was that there was only a 1" hole in the plate. The result being that when the control valve to the upper tank shut, the pump was effectively operating at its shut off point. This caused the axial shuttling which in turn destroyed the bearings.

CASE NO. 17- Large vertical pumps were cracking shafts every few weeks.

Equipment: Vertical service water pumps

Symptoms and test results: Due to the severity of the problem, underwater proximity probes were installed on one of the pumps. In addition, casing probes and torsional instrumentation was also installed. When the pump was put into operation, high levels of sub-synchronous vibration were observed. Natural frequency and mode shape measurements determined that the sub-synchronous vibration was centered around the shafts 1st lateral natural frequency. The cause of the problem was traced to a maintenance superintendent purchasing impellers from a non OEM source. The design of the impellers varied significantly from the original design. This caused high levels of turbulents that excited the shaft's natural frequency. Since non-synchronous vibration causes stress reversals, this caused the shafts to fatigue.

CASE NO. 18- Boiler feed pump would operate successfully for several months, then the running speed levels would start trending upward.

Symptoms and test results: Two feed pumps had a history of vibration problems. Following their overhauls, they would operate smoothly, but after a few months, the vibration levels would start to increase. The increase was due to higher levels of running speed vibration. When tests were finally run on the pumps, it was discovered that they were operating near a critical speed when fully loaded. This was determined when changes in speed resulted in large amplitude changes and shifts in the phase angles. A newly overhauled pump did not show these same traits. It was determined that when the seals were wearing, this reduced the Lomakin stiffening effect allowing the shaft natural frequency to drop into the operating range.

CASE No. 19- Oil whirl in a chiller unit

Equipment: A steam turbine driven chiller at a University campus heating facility.

Symptoms and test results: The inboard bearing of the turbine driving a chiller experienced repeated failures. Examination of the bearing showed that the top half had been fatigued to the point that babbit was separating from the base metal. Vibration spectra contained high levels of sub-synchronous vibration. Since the bearing was located next to the coupling and oil whirl was present, misalignment was suspected.
A series of alignment measurements were made across the coupling and it was found that there was significant movement during the first hour of operation. To determine which machine was moving, a laser was set up with a receiver mounted on an I-beam. It was determined that the chiller was moving. The root cause was determined to be the suction line on the chiller shrinking as the unit cooled. This shrinking caused the chiller to rock back. This in turn lifted the shaft unloading the turbine bearing causing it to go unstable.

CASE 20: 500 megawatt generator destroyed when phase lead insulation failed. Due to 120 Hz resonance.

Equipment: 500 megawatt 2 pole generator at coal plant

Symptoms and test results: A massive failure resulting from a phase to phase short led to testing the natural frequencies of phase leads on generators of a particular manufacturer. The testing showed that the phase leads on this style of generator normally have resonances just above 120 HZ. This is very dangerous because 120 Hz is the rate in the U.S. at which the magnetic poles pass by a stationary structure, thereby providing strong excitation at that frequency. It was discovered that the phase lead natural frequencies tend to drop as the phase leads loosen up with operating time. It was determined that yearly testing was required in order that the problems be found and corrected prior to any future failures.

CASE No. 21: Coupling lock up of nuclear steam generator feed pump turbine

Equipment: Steam turbine driven feed pump.

Symptoms and test results: A feed pump turbine experienced high levels of twice running speed vibration. The orbits indicated that the problem was misalignment. According to the plant personnel, the unit had been aligned per the specifications. When the unit was brought down for an outage, the coupling was examined. Its teeth were severely worn, the lubricant had failed and it had evidently locked up. Based upon this evidence, a study of the operating alignment was made. It was determined that the original specification was wrong. Originally the pump had been set high relative to the turbine. The final setting required that the pump be set .020" lower that the turbine. The change was due to two main factors. The first was vacuum draw down of the turbine. The second was the amount of growth of the pump.

CASE No. 22: Oil Whip in a 500 Megawatt Turbine

Equipment: 500 Megawatt Turbine

Symptoms and test results: A large steam turbine had very peculiar behavior characteristics. It would operate with no problems for months at a time. If it then had to come off line for a few hours, it could not be started back up, due to high vibration from oil whip in the fisrt Lp bearing. However, if the unit was off line for a day or so, it would start back up with no problem. It would also start up OK if it was restarted immediately after being brought off line. Such a situation has all the signs of a thermally related alignment problem. Since normal alignment equipment cannot be used on an operating turbine, a special system was developed to measure the elevation changes of the bearings. This system showed that when the vacuum was drawn on the unit, the low pressure rotor bearings dropped significantly. When this effect was combined with differential thermal shrinkage as the unit cooled down, it resulted in the first LP bearing

being unloaded enough to cause oil wirl. As the unit came up to speed, the oil whirl locked onto the rotor's first natural frequency and developed into oil whip. The solution was to install a tilt pad bearing in the first LP position. After the installation of the tilt pad bearing, the hot startup problem was eliminated.

CASE NO. 23- Large 3600 RPM Motor with a thermal vector

Equipment: 4000 Hp 3600 RPM Feed Pump Motor

Symptoms and test results: A large feed pump motor was sent out for a normal inspection and cleaning. After returning from the motor shop, it was put into operation and after 45 minutes, high vibration destroyed its bearings. It was returned to the motor shop where it was rebalanced. When it was returned to the plant, it again destroyed the bearings. It was then sent back to the manufacturer and put in a high speed pit and balanced at speed. When it was reinstalled back at the plant, it destroyed the bearings for a third time. Due to the nature of the problem, proximity probes were installed on the motor. When it was first started, the vibration was normal. However, as the motor was loaded, the vibration level increased to the point that the motion was nearly equal to the bearing clearance.
It was determined that the motor had a thermal vector. The solution was to balance the motor in the loaded condition. It operated with the thermal compromise shot installed for several years. It was discovered that the motor shop had dropped the rotor on its first visit. This had damaged the laminations causing a hot shot to develop. This hot spot on one side caused the rotor to bow as it heated up thereby producing the sensitivity to load.

CASE NO. 24: Cracked Rotor Bars

EQUIPMENT: 1800 RPM 250 Hp Service water motor

Symptoms and test results: While in operation a noticeable variation in the sound pattern was evident. The current meter also showed oscillations. Based upon these symptoms, the motor was connected to a dynamometer and spectra of the current were obtained.
The spectra which were taken at various loads showed the presence of side bands spaced at the number of poles times the slip frequency in both the current and vibration spectra. Since this is a sign of broken rotor bars, the motor was disassembled and the rotor was re-barred. Following the repairs, the sidebands disappeared and the sound and current oscillations also went away.

CASE NO. 25: Over heating of D.C. Motor

Equipment: D.C. Motor on press roll at a paper mill

Symptoms and test results: A D.C. motor had repeated failures due to overheating. In addition, the vibration levels were high at a frequency of undetermined origin.
Analysis of the current indicated that there was instability of the drive. The drive was trying to speed up, then slow down the motor at a rapid rate. This is like hitting the

accelerator then the brakes rapidly on a car. The result being high vibration and heat. The drive was retuned and both the vibration and the heat problem disappeared.

Case No. 26- Process related problem picked up on D.C. Motor

Equipment: D.C. Motor on couch roll at paper mill

Symptoms and test results: A unidentified vibration was detected on the motor driving a couch roll at a paper mill. The frequency of the vibration did not match any known source. A current spectrum was taken on the motor and it was discovered that the same frequency appeared in the current spectrum. What was occurring was that the motor was being loaded and unloaded at that particular frequency. The source of the loading oscillations was traced to the fan pump blade pass frequency. The fan pump was generating pressure pulsations that caused oscillations in the head box pressure. This in turn resulted in variations in the pulp thickness. When the thicker areas passed the vacuum rolls, the suction pulled harder against fabric. This increased tension in the fabric caused the tangential force to increase on the couch roll which increased torque demand on the motor and thus the amount of current draw.